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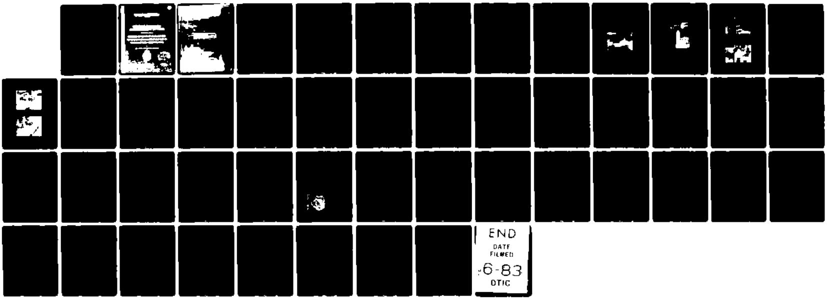

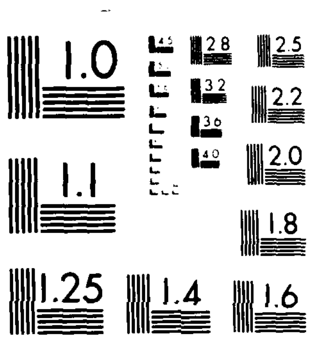
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RESEARCH AND DEVELOPMENT BRANCH

DEPARTMENT OF NATIONAL DEFENCE
CANADA

DEFENCE RESEARCH ESTABLISHMENT OTTAWA

DREO TECHNICAL NOTE 81-19

SHAKEDOWN AND PRELIMINARY CALIBRATION TESTS FOR THE FUEL ENGINE EVALUATION SYSTEM USING THE KM914A SACHS ROTARY COMBUSTION ENGINE

by

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ABSTRACT

A Fuel Engine Evaluation System (FEES) has been designed to evaluate the effects of changes in fuel composition on engine performance and operability. This report describes FEES and the tests carried out to calibrate the system and to determine its reliability. From the results obtained recommendations are made to improve the system.

FEES was designed to handle spark ignition and compression ignition research engines of power output up to 22 kW at 3500 RPM. Cold start capability testing to -40°C is also available. The engine used for the above tests was a Sachs, single rotor KM914A rotary combustion, spark ignition engine.

RESUME

Un système d'évaluation de moteur en fonction du carburant (Fuel Engine Evaluation System) (FEES) a été conçu pour évaluer les effets causés par des modifications de la composition du carburant sur le fonctionnement et la performance d'un moteur. Le rapport donne une description du FEES et des essais d'étalonnage et de fiabilité du système. A partir des résultats obtenus, des recommandations sont formulées pour améliorer le système.

Le FEES a été conçu pour des moteurs de recherche à allumage par étincelle et à allumage par compression d'une puissance d'au plus 22 kW à 3500 tr/min. Il comprend aussi des essais de démarrage à froid réalisés jusqu'à -40°C. Le moteur utilisé dans les essais mentionnés précédemment était un moteur rotatif Sachs KM914A monorotor, à allumage par étincelle.

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1.0 INTRODUCTION

A subprogram to investigate Fuel/Powerplant interaction has been initiated at the Defence Research Establishment Cttawa (DREO). One of the central objectives of the subprogram is to determine the effects of the anticipated degradation in quality of future domestic and foreign fuels on Canadian Forces powerplants. An intramural study has been implemented to complement extramural activity for the investigation of compression ignition powerplants combusting future Canadian fuels.

To monitor engine performance and operability while combusting fuels with low cetane number and/or wider boiling range, and fuels derived from tar sands blends, a Fuel Engine Evaluation System (FEES) has been designed and assembled. This report endeavours to discuss this system and the tests performed during its shakedown and preliminary calibration trials using the KM91 A Sachs Rotary Combustion Engine as a prime mover.

2.0 TEST OBJECTIVES

The tests described in this report were conducted to fulfill the following objectives:

a. to seek out equipment shortcomings or deficiencies and to determine the degree of additional equipment redundancy needed for future testing;

b. to gain preliminary insight into the equipment accuracy and behaviour in the overall system; and

c. to ascertain the degree of control necessary to hold the control parameters within test code bounds.

3.0 REPORT OBJECTIVES AND SCOPE

This report has been written to document progress of intramural research at DREO and provide the necessary background for DREO and contract personnel working in the Fuels/Powerplant Technical Subprogram. It does not attempt to describe equipment choice rationale or system design parameters but describes FEES, the tests used to fulfill the above objectives, the results obtained, the conclusions and recommendations pertinent to future testing.

4.0 THE UTILIZATION OF FEES AS A RESEARCH TOOL

FEES has been designed to identify small differences in engine performance and operability corresponding to either subtle or major changes in fuel chemistry. Initially only a comparison of the gross parameters, brake power, torque, brake specific fuel consumption, volumetric efficiency, air-fuel ratio and brake thermal efficiency will be

possible. In the near future it is planned to complement the gross parameter monitoring with the capability of combustion space monitoring for indicated power, combustion products analysis and combustion space vibrational signatures.

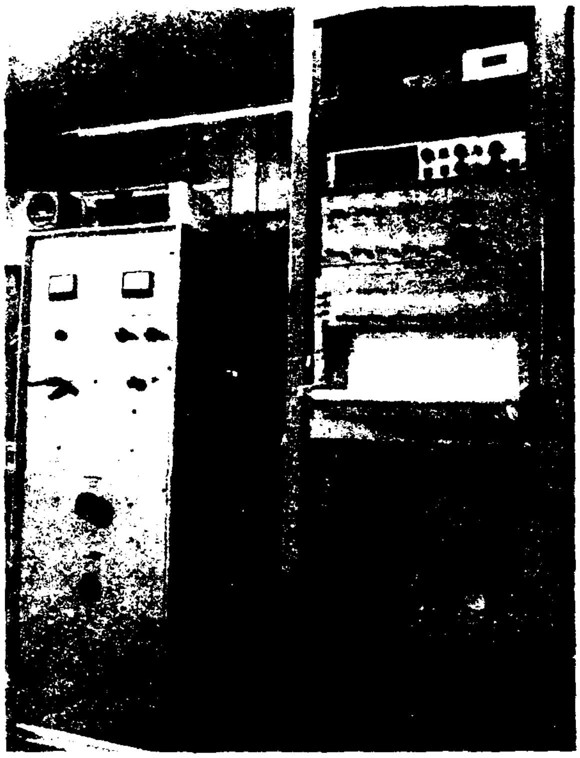
5.0 FUEL ENGINE EVALUATION SYSTEM (FEES) DESCRIPTION

FEES was designed to monitor fuel-engine performance in research powerplants of output power up to 22 Kw at speeds up to 3500 RPM over the Ottawa range of climatic conditions. Cold start capability testing to -40°C is also available. Where appropriate, the test equipment has been chosen or designed to comply with SAE and NATO fuel-engine test format standards.

Figure 1 illustrates the dynamometer end of the test bed and to the right the portable control room.

Figure 1 - [illegible]

The most basic part of the system is the test bed, a reinforced concrete slab on top of a 25 mm rubber pad. The test bed essentially rigidizes the dynamometer-engine interconnection and the rubber pad and slab dampens vibration to the building. The control room is used to house the engine-dynamometer controls, recording instrumentation and sensor readouts. Figures 2a and 2b show the engine-dynamometer controller, strip chart recorder, sensor display readouts and a data acquisition system.



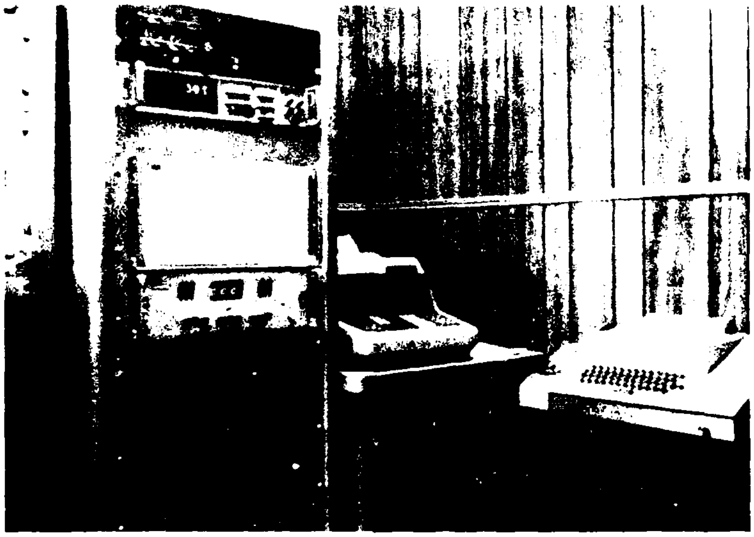


Figure [illegible] - [illegible]

For the tests of this study a Sachs KM914A rotary combustion engine was used as a calibration prime mover. Figure 3 illustrates the torque meter couplings between the engine and the DC dynamometer.

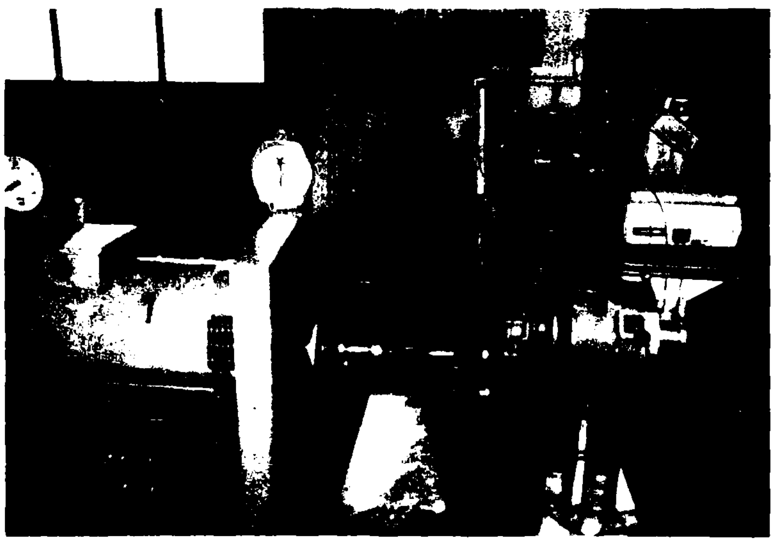


Figure 3 - Dynamometer - Torquemeter - Engine

On both ends of the torquemeter, flexible rubber couplings accommodate slight misalignment and prevent shock loads being transmitted from the engine to the dynamometer. In order to reduce power dissipation due to misalignment, an alignment jig was developed to permit periodic checks of engine/torquemeter alignment with the dynamometer.

Figure 4 a schematic drawing of the entire system, itemizes measurement locations and lists recording or readout devices. As can be observed, a number of parameters are measured twice to improve the experimental reliability.

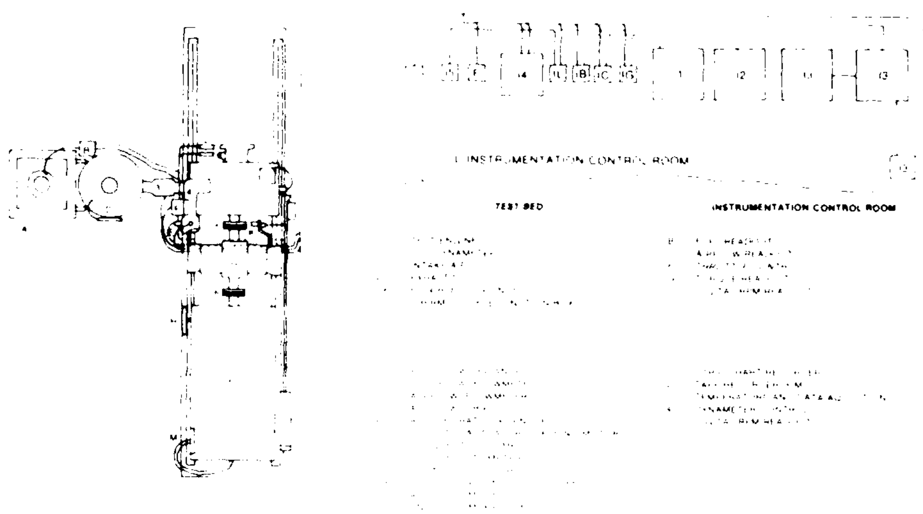
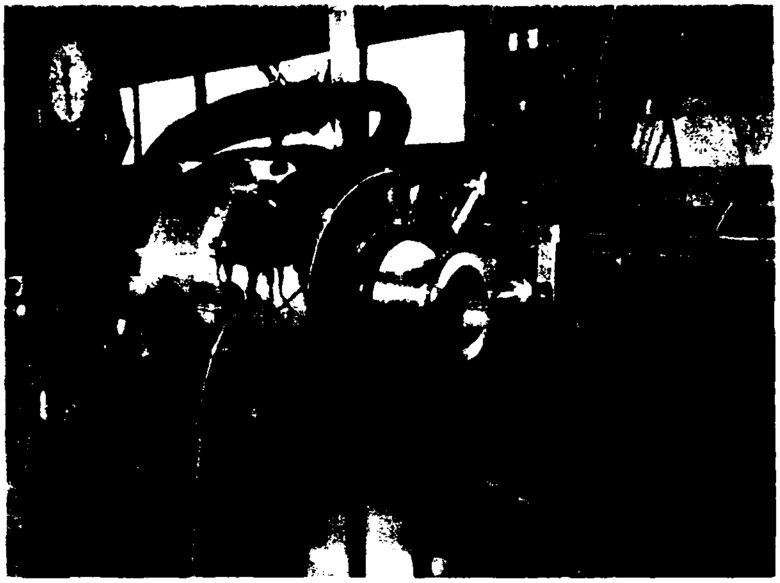


Figure 4 - [illegible] Schematic

Figures 5a and 5b indicate the sensor locations of the schematic relative to the engine-dynamometer position on the test bed. In greater detail, Appendix A lists each of the measured parameters, the symbol designated to it, and the sensing measurement device.



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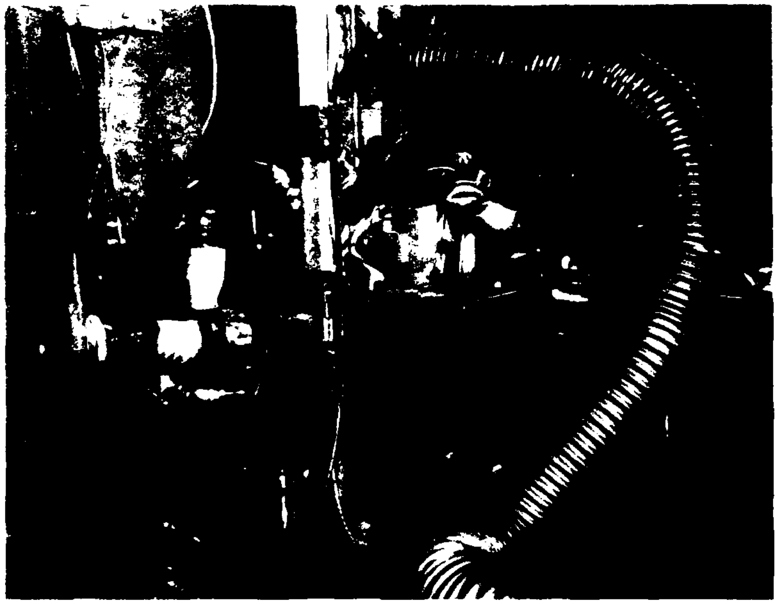

[illegible]

To permit the initiation of testing as quickly as possible, FEES had its carefully documented circuitry wired in a temporary fashion. This also facilitated the inevitable changes that this study would recommend.

To familiarize the reader with the KM914A engine, specification sheets have been inserted as Appendix B. As can be seen from the Appendix B performance curves, the engine is capable of producing 12 kW at 4500 RPM. Its peak torque 26 NM occurs at 3750 RPM and its specific fuel consumption is a minimum of 450 g/kWh at 4100 RPM. The engine is a single rotor, carburetted (Tilotson) spark ignited rotary combustion prime mover. Starting is accomplished by using the D.C. dynamometer in the motoring mode.

A refrigeration system capable of temperatures down to -40°C has been fabricated. The use of the D.C. dynamometer with the engine in a refrigerated enclosure provides suitable cold cranking capability. The proposed addition of combustion air and cooling air temperature-humidity control to -40°C should also permit low temperature engine operation for fuels testing of research type single cylinder engines.

6.0 TEST PROGRAMME

It was decided to conduct tests at three different throttle openings 50%, 75% and 100%. This was accomplished in a reproducible manner using a throttle actuator and electronic control device. At each throttle opening the dynamometer load was set to determine the operating RPM. In this fashion the RPM was varied between 2000 and 4000 RPM. For a particular test run at a fixed throttle opening and adjusted load condition, the engine was operated for approximately 5 to 15 minutes in a pretest period until temperature stabilization occurred at the exhaust manifold thermocouple. Once temperature stability was present the test run was begun and readings were recorded for a known time period of approximately four minutes. After this time period elapsed, exhaust manifold temperature was again observed. If this continued to be stable, results were calculated immediately and torque and brake power plotted to compare with previous test runs. During this post test period of two to three minutes other results were fed into the DREO site main frame computer. If the plotted data compared favourably and the temperature of the exhaust gases and other temperatures remained stable, a new load setting was effected and pretest checking for a new test run was commenced.

To minimize errors and omissions, with the large number of sensors and variety of recording or readout devices, elaborate check lists were devised. Check lists at the beginning and end of the testing day ensured proper calibration of all equipment. Pretest check lists ensured all previous data was recorded correctly and that the next data would be taken in the correct format. Due to the thoroughness of check lists, no errors or omissions resulted during the 14 tests for this study.

The results of a series of tests for a particular throttle opening led to the calculation of the performance parameters. These performance parameters outlined in the following section are treated in greater detail and their interrelationships discussed in Taylor [1], Obert [2] and Lichty [3]. From the performance parameters, performance curves can be plotted which form the basis for comparisons between test fuels and a baseline or reference fuel. In addition, the engine can be observed for its operability characteristics, i.e. how effectively or smoothly it operates with a particular test fuel. (This observation can be quantified by utilizing accelerometers mounted in various engine locations to measure combustion roughness or vibrational signature.)

7.0 GROSS PERFORMANCE PARAMETER EQUATIONS

The gross performance parameters are calculated from the measured parameters described in Table A1 of Appendix A in the following manner. Engine steady state output torque (T) is read directly from the torquemeter in (in/lbs) and converted to (NM) or calculated by

$$T = F_s \times r \text{ (NM)} \qquad (1)$$

where F_s is the dynamometer reaction force and r is the radius measured from the dynamometer centre line to the point of force measurement. The brake power (bp) is calculated using

$$bp = \frac{2\pi\ NT}{60} \text{ (Watts)} \qquad (2)$$

where N = Engine Speed (RPM)
T = Torque (NM).

The brake specific fuel consumption (bsfc) is the mass flow rate of the fuel ($\dot{m}_f$) divided by the brake power (bp).

$$BSFC = \frac{\dot{m}_f}{bp} \qquad (3)$$

Using the lower calorific value of the fuel (LCV) determined from API gravity and aniline point for the ASTM (D1405-64) test code, the brake thermal efficiency ($\eta_{\beta T}$) can be calculated from

$$\eta_{\beta T} = \frac{bp}{\dot{m}_f \times LCV} \qquad (4)$$

Another performance parameter is the cylinder mean effective pressure which can be expressed as the brake mean effective pressure (bmep).

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1 The torquemeter used also had the ability to measure instantaneous torque of the engine output shaft for monitoring combustion space energy transfer.

$$\text{bmep} = \frac{\text{bp}}{D_T \times N} \quad (5)$$

where (D_T) is the total displacement and (N) is the engine rotational speed in RPM. The bmep may be thought of as that mean effective pressure acting on the pistons which would give the measured bp if the engine were frictionless.

For air breathing engines air capacity ($\dot{m}_a$) is an important parameter. Its effect on indicated power (IP) can be seen in the following equation.

$$IP = J\dot{m}_a\ (FQ_c\eta_{TH}) \quad (6)$$

where: IP = power developed;
J = mechanical equivalent of heat;
$\dot{m}_a$ = mass flow rate of dry air;
Q_c = heat of combustion;
η_{TH} = indicated thermal efficiency; and
F = fuel/air ratio.

For SI engines, the indicated power remains proportional to air capacity, provided there is no change in fuel/air ratio or compression ratio and no departure from optimum spark timing. For these conditions, indicated thermal efficiency remains reasonably constant and this proportionality between IP and $\dot{m}_a$ has been found to hold for many types of SI engines.

To further examine the aspiration capability of an engine, it is convenient to use an expression independent of cylinder size. Such an expression is the volumetric efficiency. For ideal engines volumetric efficiency (η_v) may be defined as

$$\eta_v = \frac{m_m}{(V_1-V_2)\rho_m} \quad (7)$$

where m_m is the mass of fresh mixture of moist air (m) supplied and ρ_m is the density at the pressure p_m and temperature T_m. The quantity $(V_1-V_2)\rho_m$ is the mass of fresh mixture which would just fill the piston displacement volume at the density of the inlet system. The volume V_1 is at the piston bottom dead center (bdc) (position $-x_1$) and the volume V_2 is at top dead center (tdc) (position $-x_2$).

For real engines the volumetric efficiency can be expressed as

$$\eta_v = \frac{2\dot{m}_m}{NV_d J\rho_m} \quad (8)$$

where $\dot{m}_m$ = mass flow rate of fresh mixture (m);
N = RPM;
V_d = total displacement volume of engine;
V_d = $(x_1-x_2)Ap$ where Ap = bore area of piston; and
ρ_m = inlet density of fresh mixture (m).

The factor 2 in equation (8) arises from the engine combustion cycle. The 2 identifies a four stroke cycle where there is one complete cycle every two crankshaft revolutions. The equation would be without the 2 for a two stroke combustion cycle.

For rotary combustion engines (RCE) there has been extensive disagreement as to what constitutes the effective displacement volume and whether the engine operates in a two or four stroke cycle. Norbye [4] indicates that RCE's are now classified as following the four stroke cycle and that the Commission Sportive Internationale (CSI) has adopted a formula which rates RCE's displacement at twice the combustion chamber volume multiplied by the number of rotors. This eliminates the 2 in equation (8) and implies RCE's breathe much like a two stroke cycle powerplant.

For equation (8), it is necessary to define inlet density as the density of fresh mixture in or near the inlet port. The resulting volumetric efficiency measures the pumping performance of cylinder and valves alone. Since it is not always possible or convenient to measure ρ_m at the inlet port, density is usually measured at the engine air intake. The volumetric efficiency based on this measurement location is called the overall volumetric efficiency. For unsupercharged engines with small pressure and temperature changes in air cleaner, carburettor and inlet manifold geometry the overall volumetric efficiency does not differ greatly from the port volumetric efficiency.

The fuel/air ratio (F) is calculated from

$$F = \frac{\dot{m}_f}{\dot{m}_a} \tag{9}$$

where $\dot{m}_f$ = mass fuel flow rate
$\dot{m}_a = \rho_a \dot{Q}_a$

where $\dot{Q}_a$ = volumetric flow rate of dry air
ρ_a = density of dry air.

The air/fuel ratio (A/F) is simply

$$\frac{A}{F} = F^{-1} \tag{10}$$

It is appropriate to introduce the term equivalence ratio (Φ), which is the quantities of equation (9) and (10) normalized by the stoichiometric or chemically correct fuel/air ratio

$$\Phi = \frac{F(\text{actual})}{F(\text{stoich})} = \frac{A/F(\text{stoich})}{A/F(\text{actual})} \tag{11}$$

For octane (C_8H_{16}) F(stoich) is usually given as 0.0668.

These performance equations have been used in a sample calculation in Appendix D for test run No. RC30. The test conditions for test run No. RC30 are tabulated in Appendix C, Tables C1 and C2 as are the data of the earlier tests RC16 to RC29.

TABLE 1 - Test Data For Tests RC16 To RC30 At Standard Temperature Pressure Humidity Conditions

RUN NO.	Z (THROTTLE OPENING) (%)	N (RPM)	TORQUE (N-M)	Ma (g.sec^{-1}) V_a (ℓ.sec^{-1})	M_f (g.sec^{-1}) V_f (cc.sec^{-1})	BP (KW)	BSFC (g/KW-HR)	η_{BT} (%)	BMEP (MPa) (PSI)	ηv (%)	$F \times 10^{-2}$	F^{-1}	Φ	T_{AMB} (°C)	T_{FUEL} (°C)	T_{EXH} (°C)	RUN NO.
RC16	50	2050	19.5	8.91 7.39	.66 .94	4.18	568.4	13.26	.404 58.6	71.78	7.41	13.50	1.109	19.4	13.8	452	RC16
RC17	50	2530	22.4	11.79 9.86	.96 1.35	5.93	582.8	12.93	.464 67.3	77.53	8.14	12.28	1.218	22.8	15.9	527	RC17
RC18	50	3000	22.9	15.99 13.38	1.15 1.61	7.19	575.8	13.09	.474 68.8	88.74	7.19	13.90	1.076	20.8	14.1	581	RC18
RC19	50	3500	22.7	18.88 15.68	1.37 1.91	8.31	593.5	12.70	.470 68.2	89.16	7.26	13.78	1.087	19.8	14.6	612	RC19
RC20	50	4000	24.3	20.12 16.90	1.45 2.03	10.17	513.3	14.68	.503 73.0	84.07	7.26	13.78	1.087	22.4	15.8	678	RC20
RC21	75	2000	18.4	8.18 6.81	.65 .90	3.85	607.8	12.42	.381 55.3	67.79	8.07	12.39	1.208	16.2	17.8	443	RC21
RC22	75	2500	22.7	10.43 8.72	.91 1.27	5.94	551.5	13.69	.470 68.2	69.40	8.72	11.46	1.305	18.7	18.6	525	RC22
RC23	75	3000	23.5	15.80 13.22	1.22 1.70	7.38	595.1	12.69	.487 70.6	87.69	7.72	12.95	1.156	18.8	17.8	571	RC23
RC24	75	3500	25.4	20.03 16.79	1.52 2.11	9.30	588.4	12.83	.526 76.4	95.48	7.59	13.18	1.136	20.2	15.9	618	RC24
RC25	75	3750	25.2	24.18 19.65	1.56 2.15	9.89	567.8	13.30	.522 75.8	104.29	6.45	15.50	0.996	16.2	14.5	657	RC25
RC26	100	2000	19.8	7.48 5.83	.68 .93	4.14	591.3	12.84	.410 59.5	58.03	9.09	11.00	1.361	3.6	9.6	451	RC26
RC27	100	2500	23.5	11.30 8.98	.97 1.33	6.15	567.8	13.37	.487 70.6	70.66	8.58	11.65	1.284	5.9	7.9	537	RC27
RC28	100	3000	25.7	15.72 12.48	1.21 1.65	8.07	539.8	14.06	.533 77.3	82.77	7.70	12.99	1.153	5.1	7.0	601	RC28
RC29	100	3500	26.2	21.09 16.74	1.51 2.06	9.59	566.8	13.39	.542 28.8	95.18	7.16	13.97	1.072	3.1	6.9	634	RC29
RC30	100	4000	26.6	26.18 21.00	1.66 2.28	11.13	536.9	14.14	.551 80.0	104.49	6.34	15.77	0.949	10.0	8.1	680	RC30

8.0 PERFORMANCE RESULTS

TABLE 2

Test Data For API Gravity And Calculated Higher Heating Value

RUN NO.	Z (%)	API	HIGHER HEATING VALUE (BTU/LB)
RC (16-20)	50	65.8	20552
RC (21-25)	75	64.8	20512
RC (26-30)	100	62.0	20400
Test Altitude = 232 feet above sea level			

The performance equation data has been tabulated in Tables 1 and 2 and plotted against engine speed in the graphs of Figures 6 to 12. From the torque values of Figures 6a, a smooth curve is present for the 100% throttle opening case.

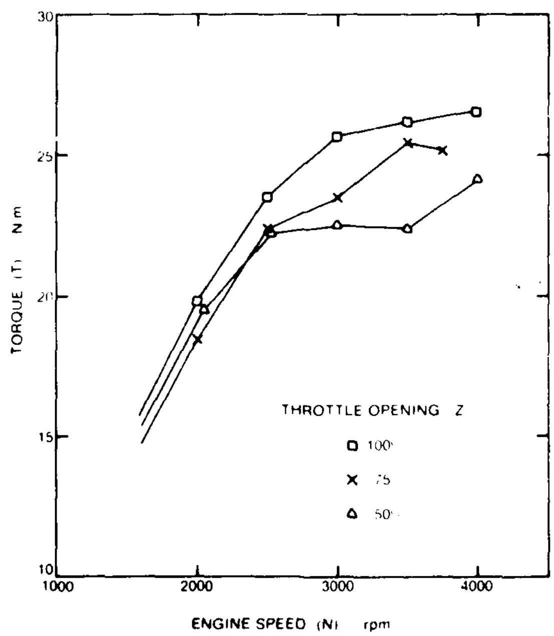

Figure 6a - Torque vs Engine Speed for the Sachs KM914A Engine

For the 75% and 50% part throttle opening curves, scatter in data was primarily due to difficulties in reading the dynamometer spring balance. Severe vibrations at certain frequencies caused erratic needle deflection, that was difficult to read accurately. These vibrations were observed to be a function of engine speed and load. Fortunately excessive vibration did not occur for any of the runs of primary importance at 100% throttle opening. It can be seen that smooth curves exist for all the air mass flow rate and fuel mass flow rate data of Figures 6b and 6c respectively. This indicates that the scatter present in Figure 6a is not engine related but dynamometer related.

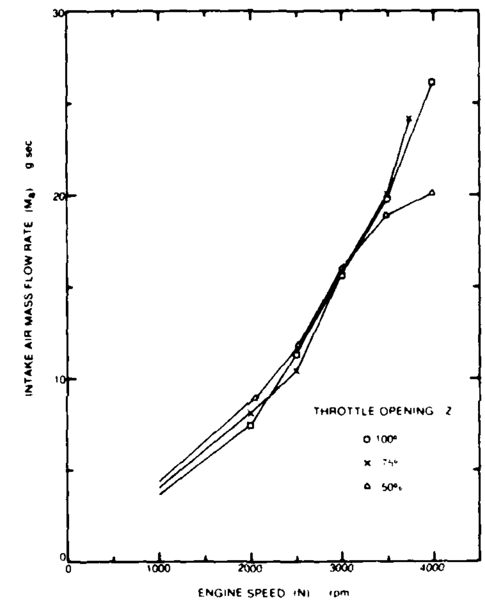

Figure 6b - Intake Air Mass Flow Rate vs Engine Speed

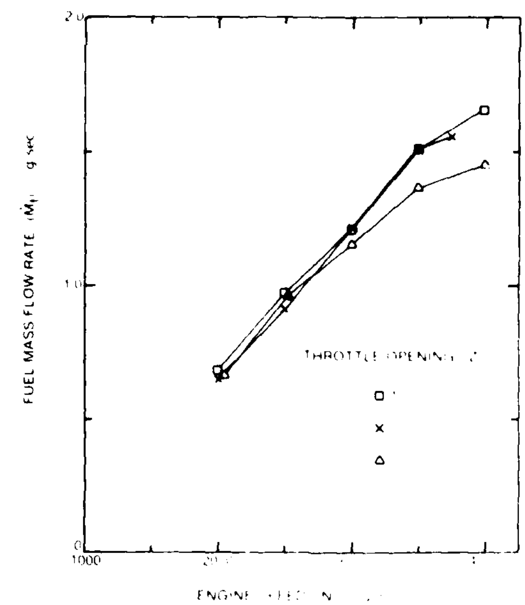

Figure 6c - Fuel Mass Flow Rate vs Engine Speed

The electric dynamometer proved to be difficult to control throughout the tests. The D.C. power to the dynamometer field coils was controlled by a potentiometer which was continually adjusted by hand to keep a standing pressure wave form generated by the combustion space pressure transducer centred on the oscilloscope screen for a given engine rotational frequency. This yielded speed control to within 15 RPM and load control for dynamometer torque of 4 NM. Engine torque is usually measured by the shaft torque meter shown in Figure 3, but due to brush problems with the torque meter, the backup spring balance was used for all of these tests.

From equation (2) the brake power (bp) values of Table 1 were generated and the smooth curves of Figure 7 were plotted. Despite minor scatter at low speeds, the crude speed/load control and the inaccurate spring balance, the bp results were encouraging. These results suggested that bp accuracy and repeatability within 1.5% for fuels testing would be possible with improved dynamometer control and use of the torque meter.

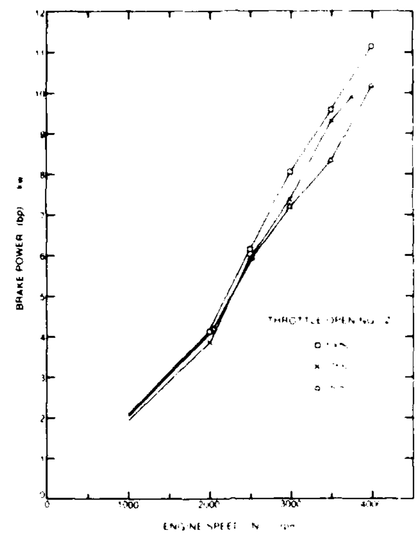

Figure 7 - Brake Power vs Engine Speed

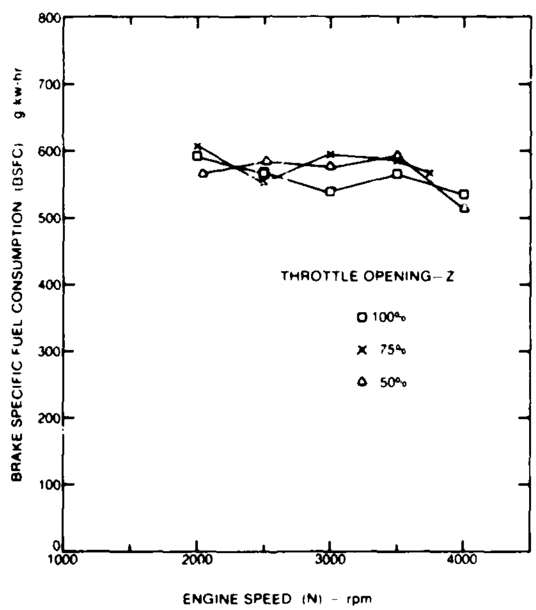

Figure 8 - Brake Specific Fuel Composition vs Engine Speed

The brake specific fuel consumption curves of Figure 8 bear out the finding of other researchers [5, 6] that full throttle and part throttle fuel consumption is poor. The curves are flatter than expected and the fuel consumption is greater than factory specifications by about 100 g/kWhr.

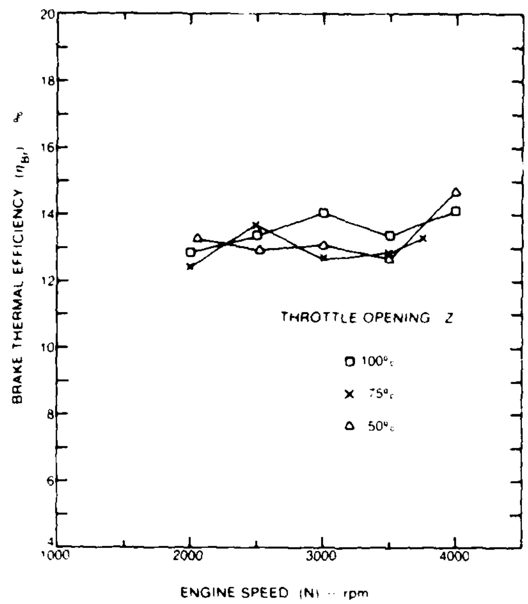

Figure 9 - Brake Thermal Efficiency vs Engine Speed

From the brake thermal efficiency curves of Figure 9, it is evident that this particular rotary combustion engine is highly inefficient. From equation (4) and Figures 6c and 7, the high fuel consumption at 3500 RPM yielded a pronounced drop in efficiency in Figure 9. It is suspected that the poor performance of the engine in terms of BSFC and η_{BT} was due to compression-combustion-expansion leakage past the rotor face and apex seals. This leakage should result in a lower brake power which was not the case. The brake power in Figure 7 compared favourably with the manufacturer's ratings. The reason for this anomaly is related to exhaust back pressure. Had the engine been tested with higher back pressure equivalent to the manufacturer's muffler, the output would have been considerably lower as would be expected with leaking seals.

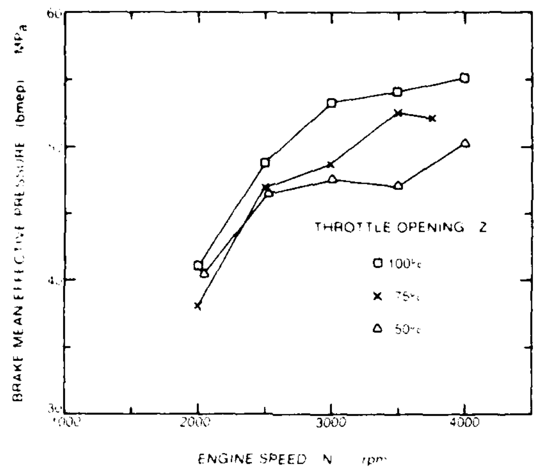

Figure 10 - Brake Mean Effective Pressure vs Engine Speed

The brake mean effective pressure of .55 mPa in Figure 10 for Z = 100% at 4000 RPM is about 2% lower than a performance map value in reference [4]. As expected, the general shape of the BMEP curves is the same as the brake power curves in Figure 7 since brake power is used to calculate bmep of equation (5).

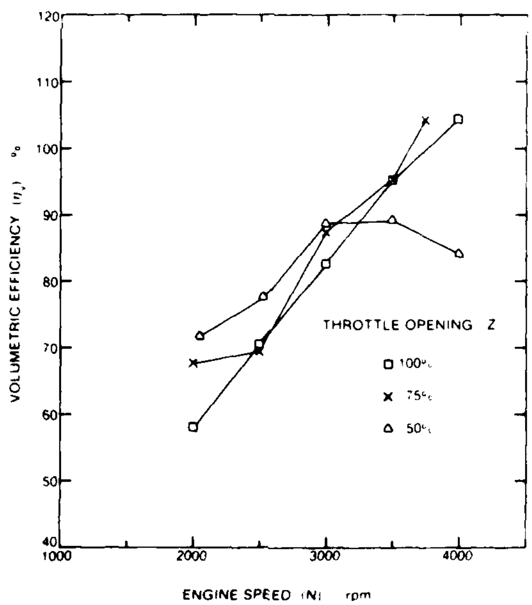

Figure 11 - Volumetric Efficiency vs Engine Speed

From Figure 11 it can be seen that, at full throttle opening, the volumetric efficiency increases linearly with engine speed and reaches a value in excess of 100%. This is typical of rotary combustion engines as they breathe more efficiently than do reciprocating engines. One might think of a rotary engine as a type of engine that lies somewhere between a reciprocating and gas turbine powerplant.

The part throttle curves of Figure 11 can best be explained by the air/fuel ratio curves of Figure 12. The full load curve is smooth without discontinuities, whereas the 75%, and 50% curves show the effects of throttling the air flow before the combustion process. This process usually leads to so called part throttle performance flat spots that are evident at 2500 and 3500 RPM.

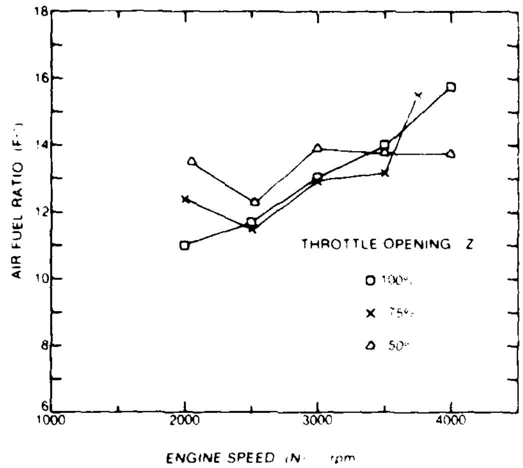

Figure 12 - Air/Fuel Ratio vs Engine Speed

The overall performance data, as tabulated in Tables 1, 2, C1 and C2, are in good agreement with the values plotted in the curves of Figures 6 to 12. The notable exception is the spring balance reading. It is planned to replace this device with torque strain gauge transducers in future testing. It was also determined through these tests that the sampling rate of the data acquisition system used for temperature measurement could not be increased to accommodate transient pressure time information.

9.0 CONCLUSIONS AND RECOMMENDATIONS

The major equipment short-coming uncovered as a result of these tests is related to the measurement of torque. It is recommended that the dynamometer spring balance reading be replaced by a strain guage brushless torque transducer. Such a transducer must be capable of reading both steady state and dynamic torque. The second equipment short-coming concerns the data acquisition system. Although suitable for temperature measurement, the system was shown to have a sampling rate inadequate for

monitoring pressure-time or other engine transients. It is recommended that a Tektronix Digital Analyzer system be considered for monitoring transient engine phenomena.

All other equipment performed as expected in either a primary or backup measurement role. It is believed that performance reading for primary measurements will be achievable to within 1 1/2 %. This will enable the system to detect small changes in performance as a result of subtle changes in fuel chemistry. It is recommended that all equipment now be hardwired into the system and coupled to a Tektronix Digital Analyzer and Fluke data logger.

In order to control the engine load provided by the D.C. dynamometer during testing a different approach than the manually controlled field coil resistor will have to be adopted. Control to within ± 10 RPM is necessary to comply with SAE specifications. It is recommended that an electronic controller be built or purchased. Such a controller will control the field current in response to a Hall effect speed sensor with a 60 tooth gear. This should provide adequate control for upcoming diesel engine fuels testing.

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APPENDIX A

PARAMETER DESCRIPTION, SYMBOL AND CORRESPONDING MEASUREMENT DEVICE

TABLE A1

Parameter Description and Measurement Device List

PARAMETER NAME	UNITS	PARAMETER DESCRIPTION	MEASUREMENT DEVICE
P_b	In Hg	Atmospheric pressure	Barometer
T_D	°F	Temperature Dry Bulb	Thermometer
T_W	°F	Temperature Web Bulb	Sling Psychrometer
Ø	%	Relative Humidity	Psychrometric chart
	mm Hg	Vapor Pressure	Vapor Pressure chart
	-	Correction factor for Humidity	Humidity Correction chart
API	-	Fuel Specific Gravity	Hydrometer
D_{NOZ}	in	Diameter of Air Drum Nozzle	Pre-calibrated
Z	°	Throttle Opening	Throttle Activator
L	-	Load	Dynamometer Load Position Knob
I	Amps	Load Current	Dynamometer Guage
V	Volts	Load Voltage	Dynamometer Guage
T	in lbs	Engine Torque	Torquemeter (Lebow)
Q_f	cc sec^{-1}	Fuel Flow rate	Positive Displ Flow Meter (Fluidyne)
Q_A	ft^3 min^{-1}	Air Flow rate	Vane Type Flowmeter (AutoTronics)
τ	min&secs	Elasped time for run	Digital Clock
V_T	cc	Total Volume of Fluid	Positive Displ Flow-meter (Fluidyne)
Θ_c	Radians	Crank Angle Position	Hall Effect Velocity Sensor (Airpax)
Θ_s	Radians	Power Take off Shaft (PTD) Position	Hall Effect Velocity Sensor (Airpax)
N_{CA}	rpm	Revolutions per minute of Crankshaft	Hall Effect Velocity Sensor (Airpax)
P_c	psi	Cylinder Pressure	Piezo Electric Pressure Transducer (Kistler)
P_A	in H_2O	Intake Air Pressure across orifice	Intake Air Flow Measur-ing (Go Power)

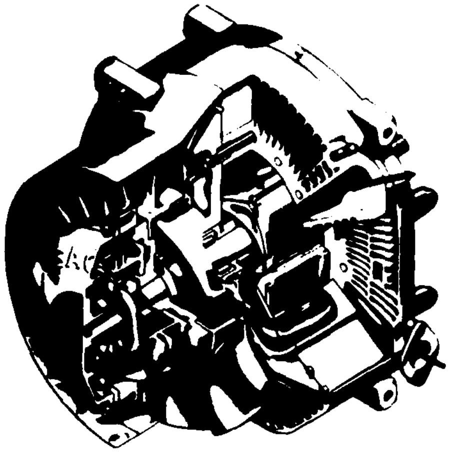
Table A1 (Cont'd)

PARAMETER NAME	UNITS	PARAMETER DESCRIPTION	MEASUREMENT DEVICE
N_{C_D}	rpm	Revolutions per minute of Crankshaft	Digital Tachometer (Go Power)
F_s	N	Dynamometer Reaction Force	Spring Balance (Go Power)
R	revs	Dynamometer cycles per unit time	Dynamometer Counter (Go Power)
M_T	kg	Mass of fuel used per unit time	Electronic Balance (Sartorius)

APPENDIX B

MANUFACTURER SPECIFICATIONS FOR THE KM914A SACHS, SPARK IGNITED, ROTARY COMBUSTION ENGINE

SACHS



Typ KM 914 A

The indicated engine output is referring to the standard equipment established by F & S for the respective engine, at barometer reading b.. = 1013 mbar and air temperature t.. 20 C for the fully run in engine (5 10 operating hours) with a tolerance of ± 5%.

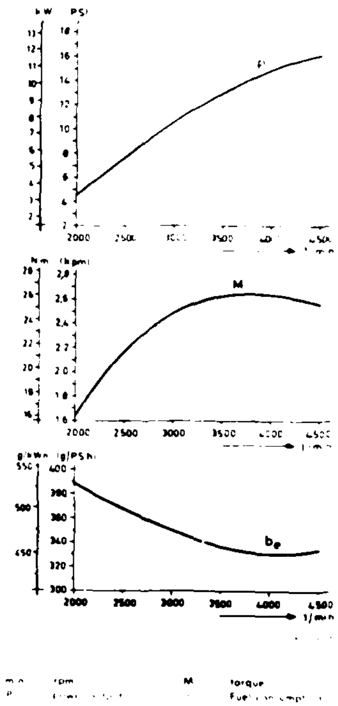
In case of a deviation from the standard equipment by addition of special parts, a change of output must be taken into consideration.

In order to determine the output in accordance with DIN 6270 (b.. = 981 mbar and t.. 20 C), the above mentioned output must be multiplied with the correction factor 0.97.

For actual continuous running (e.g. generator operation with control system) not more than 90% of the output in accordance with DIN 6270 should be used.

Standard equipment	
Construction	SACHS Wankel engine
Cooling	Air cooling by fan
Direction of rotation of engine	Anti-clockwise rotation, seen on eccentric shaft power take-off side
Chamber volume	For each chamber 303 cc
Compression	8
Output	8 kW (10.9 HP) at 3000 1/min 12 kW (16.3 HP) at 4500 1/min
Eccentric shaft bearing	3 anti friction bearings
Maximum operating speed	4500 1/min
Maximum torque	25.5 Nm (2.6 kpm) at 3500 1/min
Fuel consumption	448 g/kWh (330 g/HPh) at 4000 1/min
Engine lubrication	Two stroke mixture, special SACHS oil for Wankel engines or branded oil in accordance with operating instructions with normal fuel at a ratio of 1:50
Electrics	BOSCH magneto-generator 12 volts 40 watts
Spark advance	10 ... 12 before TDC
Breaker points gap	0.4 ± 0.05 mm
Spark plug	BOSCH W 150 M 11 S Electrode gap 0.5 mm
Carburettor	BING throttle-valve carburettor 8/25 S/135
Fuel feeding	By gravity
Air cleaner	Micronic air filter
Governor	Precision governor for 3000, 3600, 4000 and 4500 1/min Accuracy of control system ± 2.5% in accordance with DIN 1940
Muffler	Double walled with air cooling by ejector
Engine noise	of the engine loaded at n = 3000 1/min 3500 1/min 4000 1/min 78 dB (A) 80 dB (A) 82.5 dB Mean value of circumferential measurement at a distance of 7 m
Starting method	Recoil starter
Fuel tank	Capacity 6.5 litres
Weight	Appr. 32 kg
Special equipment:	BOSCH magneto-generator 12 volts 75 watts
	Fuel feed pump only to be used with BING throttle valve carburettor 8/25 S/136 (suction height 30 mm maximum)
	BOSCH starter generator 12 Volt 130 Watt or BOSCH Bendix type starter, 12 Volt 0.3 kW (necessary battery 12 Volt 18 Ah)
Mounted parts:	Single gearbox according to choice i = 1.8, i = 2.8
	Double gearbox according to choice i = 6.6, i = 10.4
	Flange shaft on disc flywheel
	Flange shaft with intermediate flange

Rotary Piston Engine



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APPENDIX C

COMPUTER PRINTOUT FOR TEST RUN #RC30

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TABLE C1

Computer Printout for Test Run #RC30

RUN BENGINE1
P1 ASSOCIATED
GROUP 17 OF :LIB.:SYS ASSOCIATED FOR 9INPUTU
* * ALLOCATION SUMMARY * *

PROTECTION	LOCATION	PAGES
DATA(00)	A000	3
PROCEDURE (01)	A800	7
DCB (10)	A600	1

FILE NAME = RC30

ATM Pressure	=	30.120	IN HG	DPB
Temp Dry Bulb	=	53.000	DEG-F	DTDF
Temp Wet Bulb	=	40.000	DEG-F	DTWF
Vapor Pressure	=	.000	MM HG	DE
Relative Humidity	=	25.000	%	DPHI
Corr. Factor For Humidity	=	1.000		DHC
API Number	=	62.000	DEG	DAPI
Diameter of Air Drum	=	.750	IN	DNOZ
Throttle Opening	=	300.000	/300	DZ
Load	=	15.000		DL
Current	=	54.000	AMP	DI
Voltage	=	177.000	VOLTS	DV
Torque (Lebow)	=	.000	IN-LBF	DTLS
Fuel Flow	=	2.260	CC/SEC	GFUEL
Air Flow	=	796.000	HZ	DAA
Time	=	400.000	MIN-SEC	DFT
Volume of Fuel	=	546.420	CC	DFVF
Period of Crank	=	.000	SEC	TETAC
Period of PTO Shaft	=	.000	SEC	TETAS
RPM (Crank), Analog	=	4000.000	RPM	NCA
Cylinder Pressure	=	.000	PSI	PCYL
Air Drum Pressure	=	.000	IN H_2O	DAM
RPM (Crank), Photo	=	4000.000	RPM	DN1
Force (Dynamo)	=	55.000	NEWTONS	DSBR
RPM (PTO), Dynamo	=	2274.000	RPM	DN2
Mass of Fuel	=	.393	KG	DFMB
No. of Revolutions	=	4528.000	REV	DREV

		(1)	(2)	(3)	(4)	(5)	
TEMPERATURES:	Initial	679.800	659.100	13.300	10.300	8.100	C
	Final	683.2	661.1	14.2	10.1	8.7	C

TABLE C2

Computer Printout for Test Run #RC30

FILE NAME = RC 30

Relative Humidity = 2.625/10.289 = 25.5%

Barometric Pressure		30.12 in of Hg		765.05 mm of Hg
Dry Bulb Temperature		11.67 C		53.00 F
Wet Bulb Temperature		4.44 C		40.00 F
Relative Humidity		25.5%		
Altitude Above SDA Level		232.00 feet		70.71 M
Vapor Pressure		2.63 mm		70.71 m
Load		15		
Current Dynamometer		54.00 ADC		
Voltage Dynamometer		177.00 UDC		
Power Dynamometer		12.82 hp		9.56 kw
Density of Fuel		.731 C/CC		
Density of Air		1.247 G/L		
Torque (Lebow Sensor)		.00 in-lbs		.00 N-M
Torque (Spring Balance)		414.21 in-lbs		46.80 N-M
Fuel Flow Rate (Fluidyne)	Volume:	2.28 CC/sec	Mass:	1.66 G/sec
		.29 CF/hr		13.21 lbs/hr
Fuel Flow Rate (Balance)	Volume:	2.24 CC/sec	Mass:	1.64 G/sec
		.28 CF/hr		13.00 lbs/hr
Air Flow Rate (Autotronics)	Volume:	21.00 L/sec	Mass:	26.18 G/sec
		2670.25 CF/hr		207.79 lbs/hr
Air Flow Rate (Manometer)	Volume:	.95 L/sec	Mass:	1.18 G/sec
		120.42 CF/hr		9.37 lbs/hr
Air/Fuel Ratio		15.73		
		15.99		
		.71		
		.72		
Brake Power		14.95 hp		11.15 kw
		.00 hp		.00 kw
Brake Specific Fuel Consumption		.87 lbs/hp-hr		.53 kg/kw-hr
		#### lbs/hp-hr		#### kg/kw-hr
		.88 lbs/hp-hr		.54 kg/kw-hr
		#### lbs/hp-hr		#### kg/kw-hr
Brake Mean Effective Pressure		80.45 lbs/sq in		
		.00 lbs/sq in		
Thermal Efficiency		14.35 %		
		.00 %		
		14.12 %		
		.00 %		
Volumetric Efficiency		104.49 %		
		4.71 %		

Table C2 (Cont'd)

Power Distribution: IP = BP+FP
20.196 = .950 + 19.179 hp

Corrected We Obtain:

Power Distribution: IP = BP+FP
20.196 = 14.952 + 5.244 hp

Corrected We Obtain:
19.2 = 13.9 + 5.2

TC = 386.0193 in-LBF

BMEPC = 74.9725 psi

APPENDIX D

GROSS PERFORMANCE PARAMETER CALCULATIONS

FOR TEST RUN #RC30

The sample calculations performed for this Appendix are for test run #RC30. The test conditions are listed as the input to a computer program in Appendix C, Table C1. The results of the program are listed in Table C2. These results can be compared with the sample calculations for this Appendix. For brevity, only one set of calculations has been submitted. These calculations use data from the instrumentation considered to be the most accurate. For instance, calculations of the air flow are based on the air turbine rather than the less accurate pressure drop across the orifice.

The gross performance equations used in the calculations are discussed in Section 7.0 in the main body of the report.

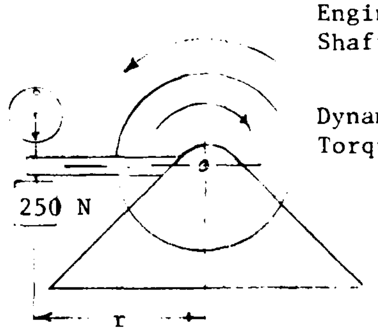

Figure D1 - Dynamometer Torque Measurement

The torque in equation (1)

$$T = F_s \times r \qquad (1)$$

is calculated by finding the true rotational force opposing engine shaft rotation. This can be found by merely subtracting the dynamometer spring indicator dial reading (F_D) from the equilibrium condition of zero torque i.e. 250N of the counter weight.

$$F_s = 250N - F_D$$

From the computer printout in Appendix C for test RC30, the input section, Table C1 lists the dynamometer dial indicator force reading (F_D) as 55N.

$$F_S = (250 - 55)N = 195N$$

The force arm radius (r) in equation (1) and Figure D1 is 0.240m. Thus torque is

$T = 195N \times 0.24m$ $\qquad$ 39.37 in = 1 m

$= 46.8$ Nm (414.2 in lb) @ 2275 RPM $\qquad$ 4.4482 N = 1 lb_f

The brake power (bp) can be calculated by substituting this value of T and the Dynamometer RPM from Table C1 in equation (2). The Dynamometer RPM of 2275 is used rather than the engine RPM of 4000 because the torque is measured on the dynamometer side of the 1.758/1 gear reduction unit. The torque before gear reduction would be

$T = 46.8Nm/1.758 = 26.6$ Nm @ 4000 RPM

$$bp = \frac{2\pi NT}{60} \qquad (2)$$

$$= \frac{2\pi rad}{rev} \times \frac{2275\ rev}{min} \times \frac{1\ min}{60\ sec} \times 46.8\ Nm\ (\frac{1\ watt}{1Nm/sec})$$

$$= (\frac{2\pi}{60} \times 2275 \times 46.8)\ watts$$

$$= 11{,}149.5\ watts\ or\ 11.15\ Kw\ (14.95\ Hp)$$

The mass fuel flow rate ($\dot{m}_f$) can be calculated by using from Table C1 the mass of fuel 393 Kg over the test period of 4.0 min (240 secs).

$$\dot{m}_f = \frac{393}{240\ sec} = 1.64\ g/sec$$

The brake specific fuel consumption (bsfc) of equation (3) can now be calculated

$$bsfc = \frac{\dot{m}_f}{bp}$$

$$= \frac{1.64\ g.sec^{-1}}{11.15\ Kw} \times \frac{3600\ sec}{1\ hour}$$

$$= 529.5\ g/Kwh.$$

It is n generally accepted that the lower calorific value (LCV) be used in determining brake thermal efficiency (equation (4)). However, for the purpose of these shakedown tests it was considered unnecessary to carry out aniline point tests for each fuel batch according to ASTM (D611-77). Instead of using the conventional means of calculating LCV, ASTM (D1405-69), the modified Sherman-Kropff equations were used to determine the higher heating value (HHV).

$$HHV = 18{,}320 + 40\ (API-10)^{*}\ in\ \frac{BTU}{lb}\ \ (for\ gasoline)$$

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*Journal of the American Chemical Society, Vol 30, No 10, p 1626.

This is an approximate HHV equation where only the API gravity is needed, ASTM (D1298-67). HHV is usually about 6 to 9% higher than the LCV. This will tend to reduce the brake thermal efficiency. As Table C1 indicates the API gravity was experimentally determined in accordance with ASTM (1298-67) and found to be 62.

$$HHV = 18{,}320 + 40\ (62-10)$$

$$= 18{,}320 + 2{,}080$$

$$= 20{,}400\ \frac{BTU}{lb}\ (47{,}405\ \frac{KJ}{Kg})$$

The brake thermal efficiency η_{BT} may now be calculated using equation (4) with the HHV revision.

$$\eta_{BT} = \frac{bp}{\dot{m}_f \times HHV}$$

$$= \frac{11.15\ Kw}{1.64\ g.sec^{-1} \times 47{,}405\ J/g}$$

$$= \frac{11.15 \times 10^3\ w}{1.64\ g.sec^{-1} \times 47{,}405\ \frac{w.sec}{g}}$$

$$= 14.34\%$$

To calculate the brake mean effective pressure (bmep) from equation (5) an equivalent displacement term must be used. For the purposes of these tests the manufacturers equivalent displacement of 303 cc (18.49 in^3) per chamber will be used.

$$bmep = \frac{bp}{\frac{\text{Total Displacement}}{\text{revolution}} \times \frac{\text{revolutions}}{\text{min}}}$$

$$= \frac{11.15 \times 10^3\ N\cdot m\cdot sec^{-1}}{\frac{303\ cm^3}{rev} \times \frac{4000\ rev}{min} \times \frac{1\ min}{60\ sec} \times \frac{1\ m^3}{10^6 cm^3}}$$

$$= \frac{11.15 \times 10^6}{303 \times \frac{4}{60}}$$

$$= \frac{11.15 \times 10^6}{20.4}$$

$$= .552 \times 10^6\ \frac{N}{m^2}$$

$$= .552\ MPa\ (79.33\ psi)$$

In order to calculate the overall volumetric efficiency (η_V) in accordance with equation (8), the density of the intake air mixture of water vapour with dry air must be found. As mentioned earlier if the volume flow rate ($\dot{V}_m$) is known for equation (8) the calculation of ρ_m is no longer necessary. The flow device used for this study had the option of using either $\dot{V}_m$ or $\dot{M}_m$ for a test run and both were alternately used. Since the reading for $\dot{M}_m$ has to be verified ρ_m will be calculated.

To calculate air-vapour mixture density a number of empirical equations exist and for atmospheric pressures between .9 and 1.1 bar a psychrometric chart is accurate to ± 3%. For completeness and benchmark purposes a fundamental approach has been used here. English units have been used for the calculations with final air densities being converted to metric units. For a more complete discussion of the following subject it is suggested the reader refer to any classical thermodynamics text.

A temperature entropy (t-s) diagram can be drawn as in Figure D1 where the environmental conditions for test RC30 are shown. The dry bulb temperature (t_d = 53°F) and the wet bulb temperature (t_w = 40°F) are located for a barometric pressure (p_m = 30.12 in of Hg).

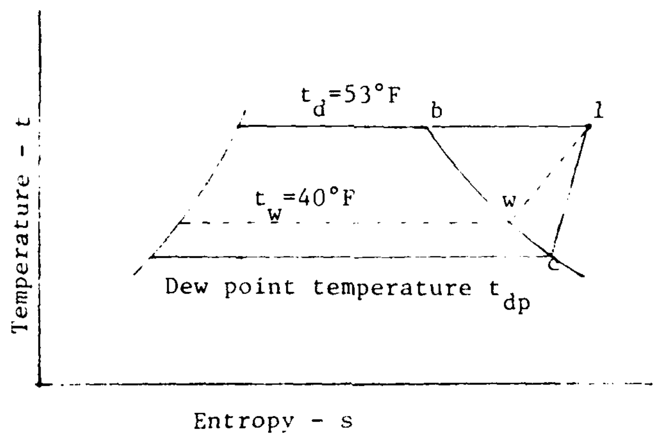

Figure D2 - Temperature Entropy Diagram for Test RC30

For an adiabatic saturation, irreversible process an energy balance for Figure D3 yields

$$h_{ad} + \omega_d h_{vd} + (\omega_w - \omega_d)h_{fw} = h_{aw} + \omega_w h_{vw} \qquad (1\text{-D})$$

where $\omega_d = \dfrac{\text{lb of vapour}}{\text{lb of dry air}}$ and evaporation = $\omega_w - \omega_d$

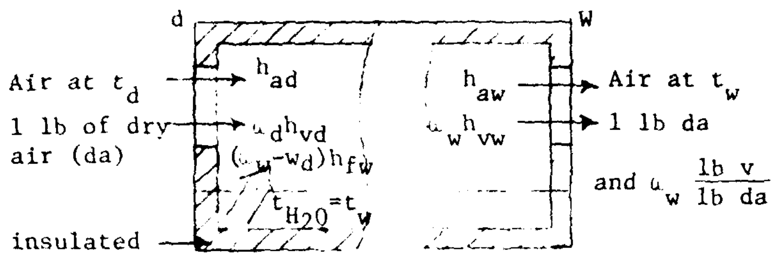

Figure D3 - Adiabatic Saturation Process

The enthalpy of evaporation symbol

$$h_{fgw} = h_{vw} - h_{fw}$$

for the exit state equation 1-D may be solved for the humidity ratio (ω_d) of the entering air.

$$\omega_d = \frac{\omega_w h_{fgw} + h_{aw} - h_{ad}}{h_{vd} - h_{fw}} \left(\frac{\text{lb of vapour}}{\text{lb of dry air}}\right) \qquad (2\text{-D})$$

where $h_{aw} - h_{ad} = C_p \Delta t_a$

and $t_a = t_w - t_d$, when $C_p = .24$ BTU/lb-°R

From thermodynamic steam tables for $t_w = 40°F$, $t_d = 53°F$ and $p_m = 30.12$ in Hg

$$h_{fgw} = 1071 \frac{\text{BTU}}{\text{lb}} \qquad h_{fw} = 8.027 \frac{\text{BTU}}{\text{lb}}$$

$$h_{gb} \simeq h_{vl} = 1084.7 \frac{\text{BTU}}{\text{lb}}$$

At $t_w = 40°F$ the partial pressure of the steam is $p_{vw} = .12163$ psia and from steam tables $v_g = 2445.8$ ft^3/lb or $\rho_{vw} = 1/2445.8 = 4.089 \times 10^{-4}$ lb/ft^3

$$p_m = 30.12 \text{ in of Hg} \times \frac{.491 \text{ psia}}{1 \text{ in of Hg}} = 14.78 \text{ psia}$$

The pressure of the air at saturation state w is

[illegible] 14.[illegible] psia

[illegible] may be calculated using the perfect gas assumption

$$\rho_{aw} = \frac{p_{aw}}{R_a T_a} = \frac{(14.6[illegible])(144)\ (lb\ in^{-2})\ (in^2\ ft^{-2})}{53.3(500\ R)\ ft\ lb\cdot(lb°R)^{-1}\ °R}$$

$$= .07925\ lb/ft^3$$

The humidity ratio after saturation state w may now be found.

$$\omega_w = \frac{\rho_{vw}}{\rho_{aw}} = \frac{4.089 \times 10^{-4}}{.07925} = 5.159 \times 10^{-3}\ \frac{lb\ v}{lb\ d_a}$$

The $h_{aw} - h_{a1}$ relationship for equation (2-D) may be found where the state 1 subscript replaces subscript d.

$$h_{aw} - h_{a1} = -C_p(t_1 - t_w) = -0.24(53-40)$$

The humidity ratio ω_1 of the original air from equation (2-D) can now be calculated

$$\omega_1 = \frac{\omega_w\ h_{fgw} + h_{aw} - h_{a1}}{h_{v1} - h_{fw}}$$

$$= \frac{(5.159 \times 10^{-3})\ (1071) - (0.24(53-40))}{1084.7 - 8.027}$$

$$= 2.234 \times 10^{-3}\ \frac{lb\ v}{lb\ d_a}\ .$$

The partial pressure of the vapour is from Carrier's equation

$$p_{v1} = \frac{\omega_1\ p_m}{0.622 + \omega_1} \qquad (3\text{-D})$$

$$p_{v1} = \frac{(2.234 \times 10^{-3})\ (14.78)}{(0.622) + (2.234 \times 10^{-3})}$$

$$= 5.289 \times 10^{-2}\ psia.$$

At t_d = 53°F the saturation pressure from steam tables is p_{v_b} = .19883 psia. The relative humidity may now be calculated

$$\phi = \frac{p_{v1}}{p_{v_b}} = \frac{5.289 \times 10^{-2}}{.19883} = .266$$

or $\quad = 26.6\%$.

From steam tables a dew point temperature $t_{dp} \approx 21°F$ may be interpolated for $p_{v_1} = .05289$ psia. The density of the mixture is the density of the constituents $\rho_m = \rho_a + \rho_v$ and the partial pressure for dry air in state 1 is

$$p_{a1} = p_m - p_{v1} = 14.78 - .05289$$

$$= 14.73 \text{ psia}$$

and the density is

$$\rho_{a1} = \frac{p_{a1}}{R_a T_a} = \frac{(14.73)(144)}{(53.3)(513)} = 7.756 \times 10^{-2} \text{ lb/ft}^3$$

Since $\emptyset$ can also be written as $\emptyset \approx \frac{p_{v_1}}{p_{v_b}}$, the density of the vapour at 1 for v_{g1} is

$$\rho_{v1} = \emptyset\, \rho_{v6} = \frac{\emptyset}{v_{g_1}} = \frac{.266}{1534.8} = 1.733 \times 10^{-4} \text{ lb/ft}^3$$

and hence the density of the mixture can be found

$$\rho_m = \rho_{a1} + \rho_{v1} = 7.756 \times 10^{-2} + 1.733 \times 10^{-4}$$

$$= 7.773 \times 10^{-2} \text{ lb/ft}^3$$

From Table C2 for test RC30 the volumetric flow rate ($\dot{V}_m$) of fresh mixture (m) was

$$\dot{V}_m = 2670 \text{ ft}^3 \cdot \text{hr}^{-1}$$

The mass flow rate of mixture can be written as

$$\dot{m}_m = \rho_m \dot{V}_m = 2670 \text{ ft}^3 \cdot \text{hr}^{-1} \times 7.773 \times 10^{-2} \text{ lb ft}^{-3} \times \frac{1 \text{ hr}}{3600 \text{ sec}}$$

$$= 5.76 \times 10^{-2} \text{ lb sec}^{-1}$$

$$= 5.76 \times 10^{-2} \text{ lb sec}^{-1} \times \frac{454 \text{ g}}{1 \text{ lb}}$$

$$= 2.62 \times 10 \text{ g} \cdot \text{sec}^{-1} \quad .$$

This value compared favourably with the mass flow rate readout of Table C2 26.18 $g \cdot sec^{-1}$. The above calculation is the only means of checking the mass flow rate meter's readout which has been obtained by aid of a temperature compensated thermister at the meter's turbine throat.

The mass flow rate of the mixture may now be used to calculate the overall volumetric efficiency in equation (8). For the calculations of this report the displacement volume (V_d) was not multiplied by two as was suggested in Reference [2].

$$\eta_v = \frac{\dot{m}_m}{N V_d \rho_m}$$

$$\dot{m}_m = 2.62 \times 10\ g \cdot sec^{-1} \qquad N = 4000\ RPM$$

$$\text{or} \quad (5.76 \times 10^{-2}\ lb \cdot sec^{-1}) \qquad \rho_m = 7.773\ lb/ft^3$$

$$V_d = 18.49\ in^3$$

$$\eta_v = \frac{5.76 \times 10^{-2}\ lb\ sec^{-1} \times 60\ sec}{4000 \frac{rev}{min} \times 18.49\ in^3 \times 7.773 \frac{lb}{ft^3} \times \frac{1\ ft^3}{1728\ in^3}}$$

$$= 103.8\%$$

The fuel/air ratio of equation (9) may now be calculated

$$F = \frac{\dot{m}_f}{\dot{m}_a}, \quad \dot{m}_f = 1.64\ g \cdot sec^{-1}$$

$$\dot{m}_a = \rho_a Q_a, \text{ where } Q_a \simeq Q_m$$

$$\dot{m}_a = 7.756 \times 10^{-2} \frac{lb}{ft^3} \times \frac{2670\ ft^3}{hr} \times \frac{1\ hr}{3600\ sec}$$

$$= 5.752 \times 10^{-2} \frac{lb}{sec} \times \frac{454\ g}{1\ lb}$$

$$= 2.612 \times 10\ g \cdot sec^{-1}$$

$$F = \frac{1.64\ g \cdot sec^{-1}}{2.612 \times 10^{+1}\ g \cdot sec^{-1}} = 6.279 \times 10^{-2}$$

or the air/fuel ratio of equation (10) is

$$\frac{A}{F} = F^{-1} = 15.93$$

and the equivalence ratio of equation (11) is

$$\Phi = \frac{F(\text{actual})}{F(\text{stoich})} = \frac{6.279 \times 10^{-2}}{6.68 \times 10^{-2}}$$

$$= .9399 \ .$$

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13 ABSTRACT

A Fuel Engine Evaluation System (FEES) has been designed to evaluate the effects of changes in fuel composition on engine performance and operability. This report describes FEES and the tests carried out to calibrate the system and to determine its reliability. From the results obtained recommendations are made to improve the system.

FEES was designed to handle spark ignition and compression ignition research engines of power output up to 22 Kw at 3500 RPM. Cold start capability testing to -40°C is also available. The engine used for the above tests was a Sachs, single rotor KM914A rotary combustion engine.

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